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CENTRIFUGAL COMPRESSOR SIDESTREAM SECTIONAL PERFORMANCE PREDICTION METHODOLOGY

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ABSTRACT

The objective of this paper is to demonstrate the sensitivity of sidestream sectional performance curves to the volume flow ratio, sidestream loss coefficient and section inlet conditions. A model to determine sectional performance is outlined. This model is then used to conduct a sensitivity analysis, which is based on current ASME PTC-10 guidelines. The model is validated by comparing test results to those predicted by the model. The model is used to show the sensitivity of test results based on the mentioned parameters. The subsequent variations are also compared with current API-617 guarantees and based on this study additional guidelines are recommended to be added to the ASME PTC-10 code. The improved prediction and testing methodology is expected to provide more accurate sectional curves to clients for sidestream applications.

INTRODUCTION

Recent advances in the accuracy of analytical tools utilizing Computational Fluid Dynamics (CFD) have made it easier for Original Equipment Manufacturers (OEMs) to predict performance of advanced turbomachinery components through

CFD analyses. However, for heavy hydrocarbon applications, such as compressors used in Liquefied Natural Gas (LNG), ethylene or gas-to-liquid facilities that typically operate at higher Mach numbers and use higher flow coefficient stages, the end users continue to heavily rely on testing to confirm performance of these compressors. Centrifugal compressors used for these applications generally operate with relatively narrow flow maps compared to light hydrocarbons.

Several process applications such as chemical process plants or refrigeration service require centrifugal compressors that have single or multiple incoming flows which mix with the internal core flow of the compressor. This mixing serves to complicate the performance prediction process as the pressure, temperature and flow conditions at each of these incoming flow streams must be met within stringent pressure and power tolerances. For this reason, it is imperative for OEMs to produce accurate flange to flange predictions.

This study concentrates on the methodology used to predict compressor performance and its application in predicting performance for any sections with sidestreams. Reference test results were provided by a small impeller diameter test vehicle 0.3-0.5m (12'-20") equipped with an array of internal instrumentation. Sufficient instrumentation was installed to allow assessment of the entire compressor as well as individual components or combinations of components. This included instrumentation at critical locations within the sidestream components, to permit an assessment of the losses through the sidestream element.

Acceptance testing of industrial centrifugal compressors is governed two specifications API-617 (2009) and ASME Performance Test Code (PTC-10 1997). The ASME code provides stringent guidelines that must be met to achieve test similitude for a Type II compressor test. The primary parameters for similitude are impeller tip Mach number and the ratio of volume flow entering and discharging the compressor (volume ratio). For applications where sidestream flows are present the code stipulates limits on the ratio of flow entering the sidestream versus the core flow exiting the upstream section. The acceptable variation in volume flow ratio for intermediate sections of a compressor with multiple sidestreams is ±10% (±5% for first section). The effect of varying flow ratio in a sidestream has a potentially significant impact on the flange total pressure and hence the sectional performance. The overall guarantees outlined in API-617 require the compressor power to be maintained within ± 4%. For larger power applications, clients often request power tolerances of ±2%. The requested sidestream pressure tolerance is not specified in either document but is usually on the order of ± 2%.

The objective of this paper is to demonstrate the sensitivity of sectional performance curves to the flow ratio at each sidestream (ratio of core flow to sidestream flow) and to the inlet conditions. Also discussed is the flow phenomena in this area of the machine, quantification methods, validation process and correlated models used to predict sectional performance curves at specified conditions. The paper will outline these changes in detail and provide an explanation on how these changes can be achieved during testing. In addition to improving performance predictions over the compressor operating range prior to actual testing, this improved prediction and testing methodology will also allow the OEM to provide more accurate sectional curves to clients for sidestream applications.

BACKGROUND

When selecting and designing a compressor, the OEM must be able to predict overall performance of the unit from inlet flange to discharge flange. While this is relatively simple for machines where all stages are placed serially one after another, it becomes more complex when one or more intermediate flows are introduced between stages and merged into the main flow. In such cases, a sound understanding of the changes in the static and total pressures in these mixing zones is important in order to determine the sidestream flange total pressures.

For compressors having flows added (or removed) at intermediate sections, ASME code (PTC-10 1997) defines a section from one flange to the next flange. In order to determine sectional performance, section 3.5.5 of the code states that "The sectional head, efficiencies, and pressures are defined flange to flange. The only internal measurements needed are the sectional discharge temperatures for computing the mixed temperature conditions and sectional performance". Based on this definition, a section is not a true thermodynamic control volume as it does not satisfy the mass-energy balance principle. This implies that when the sidestream losses are attributed to the downstream section, the upstream section will appear to have a fictitiously high performance and the downstream section will appear to have an erroneously low performance. Hence, when these sectional curves are provided to the end user, they are often viewed with skepticism. Failure to understand the interaction of the flange losses with the internal compressor performance leads to incorrect conclusions on the relative performance of individual sections of a compressor system.

Compressor OEM's have completed significant work in recent years to improve their understanding of the flow in sidestreams and sidestream mixing section as well as the impact of the merged stream on the downstream impeller. In most cases, the portion of the sidestream from the flange connection to the "mixing section" (location where the incoming sidestream flow merges with the core flow) is very similar to a radial compressor inlet. There has been significant work reported in the literature on radial inlets. For example, Flathers, et al. (1994) compared several geometric variations using Computational Fluid Dynamics (CFD) and measured test results.

Additional studies, which included the upstream return channel and the complete sidestream geometry, have been completed to accurately model the flow where the core flow and sidestream flow merge. While this location is sometimes called the mixing section, the flows do not completely mix before entering the impeller. The works of Sorokes et al. (2000, 2006) and the prior referenced work on radial inlets have led to a greater comprehension of the complex flows at the sidestream mixing location. However, these works did not address the flow physics that determine the static pressure in the mixing section, which, in turn, sets the flange pressure level. The study by Hardin (2002) describes a one-dimensional method that determines how the flow at the mixing location and, therefore, the local static pressure is impacted by the local flow curvature.

A study by Koch et al. (2011) was conducted to predict the downstream total pressure and compare the analytical results to measured test data on two different geometries. This paper recommended a revised methodology to accurately predict mixing zone conditions by taking the local curvatures into account. It was validated using test results and was found to be within ± 1.5% error for the total pressure at the flange.

The intention of this paper is to continue that study by applying it to sectional performance and further evaluating the effect of varying flow per the ASME code (PTC-10 1997) guidelines, on the sectional efficiency, head coefficient and sidestream flange pressure.

SECTIONAL PERFORMANCE SENSITIVITY

The sensitivity of sectional performance to flange losses can be best understood through a simplified example. Consider a fictitious compressor with three sections with one impeller per section (Figure 1). Each section consists of a single stage with the same design flow coefficient and tip Mach number. The stage has an arbitrarily selected peak efficiency of 84%. For the purposes of illustration the internal performance of each stage is assumed be identical. There are two sidestreams in the machine, the first entering after the first stage and the second enter after the second stage. The same gas mixture is assumed for all incoming flow streams. The sidestream flow temperature is equated to the previous stage discharge temperature. Each sidestream flow is adjusted such that upon merging with the core flow, the flow entering the subsequent stage is at its design flow coefficient. Consequently each stage is operating at design point.

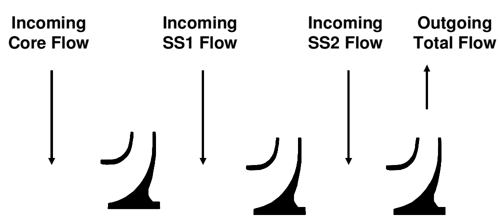

Figure 1. Layout of Example Sidestream Machine

The geometry of the mixing sections would be sized to have the same local velocity and identical curvature thus having the same static and total pressure. Without any losses, the pressure at the flanges would be identical to the pressure at the stage inlet, thus the stage performance (internal) would be identical to the sectional (external) performance for all stages/sections (Figure 2).

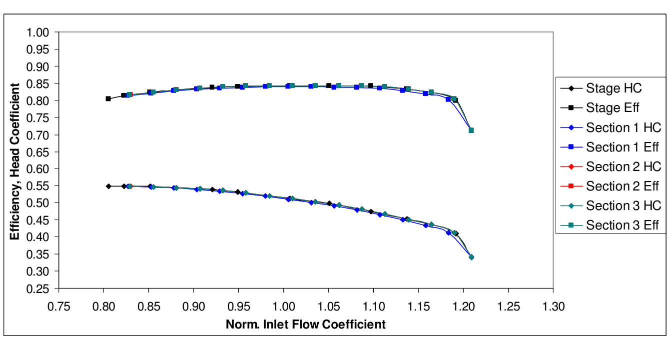

Figure 2. Performance of Example Machine with no Losses

In reality, there are pressure changes that occur from a sidestream flange to the inlet of the next stage, which depend primarily on three factors:

1. The frictional losses due to sidestream geometry (flange, plenum and mixing section geometries).
2. The static pressure changes due to local curvature and mean velocity
3. The flow ratio between the sidestream and core flows.

Flow ratio, per the ASME code (PTC-10 1997), is defined as the ratio of the volume flow rate at the sidestream flange to the volume flow rate at the exit of the preceding stage. Kolata and Colby (1990) define a flow function which is determined slightly different from the ASME code flow ratio. For all the test cases presented in this paper, for any variation in the flow ratio, the corresponding variation in the flow function is near identical (within ±0.1% of each other). Flow ratio defined as volume flow rate at the exit of preceding stage to the volume flow rate at the sidestream flange will be used throughout this paper.

Effect of Sidestream Geometry on Sectional Performance

For a given flow ratio and a given sidestream geometry, the loss coefficient from the sidestream flange to the stage inlet is a fixed value. The loss coefficient is defined by Equation 2:

$$LC = \frac{Pt_{SS} - Pt_{STG}}{Pd_{SS}} \qquad (2)$$

Based on the above equation, a higher total pressure at the flange than at the stage inlet implies a positive loss coefficient. Conversely, a lower total pressure at the flange than at the stage inlet implies a negative loss coefficient, and is not a violation of basic laws of physics. Since flow direction is governed by static pressure at each location, it is theoretically possible to have a negative loss coefficient and this often occurs. A negative loss coefficient implies that the total pressure at stage inlet is higher than that at the sidestream flange, but the static pressure at the stage inlet is lower than that at the flange.

The impact on flange to flange performance is illustrated in the following four scenarios. For these scenarios, the sidestream loss coefficient is systematically varied to show the impact on flange to flange performance (Table 1).

Scenario	SS #1	SS#2
1	4.0	4.0
2	4.0	-4.0
3	-4.0	4.0
4	-4.0	-4.0

Table 1. Flange Loss Scenarios

Scenario 1 assumes both sidestreams in the example case are designed in an identical manner and there exists a loss coefficient of 4 for each sidestream (the value of 4 was taken arbitrarily for this hypothetical case). Figure 3 shows the resulting performance curves for each section versus the internal performance curve. For this scenario the first section has the highest performance, followed by the second section and finally the third section with the lowest performance.

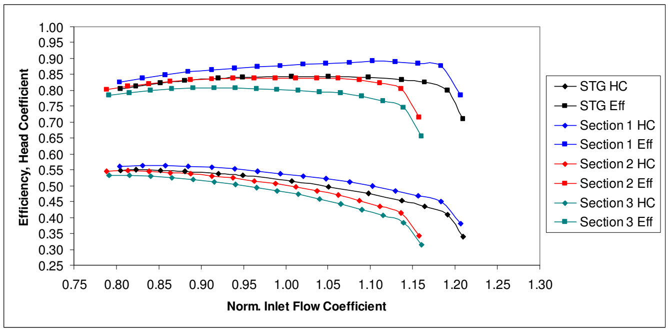

Figure 3. Sectional Performance: LC = 4 for both SS

In the case of the first section, the flange pressure at the first sidestream is higher than the stage exit pressure. When performance is determined flange to flange, the higher flange pressure is seen as extra “pseudo” work done by the stage (since this is an inconsistent thermodynamic volume), resulting in a sectional efficiency which is higher than the stage efficiency. On the contrary, the third section inlet flange pressure is higher than the stage inlet so the sectional performance is poorer than the stage performance. Since performance of section 2 is calculated based on a higher pressure at inlet and discharge flange its performance falls in between that of section 1 and 2. Another key point is that the performance curves of section 2 and 3 are shifted to reduced flow, relative to the stage curve, due to the lower higher inlet pressure which results in a reduced inlet volume flow.

Now consider scenario 2, where the first sidestream has a positive loss coefficient (4) and the second sidestream has a negative loss coefficient (-4). In this scenario the first section performance will remain the same (Figure 4). However, the third section will show a sectional performance that is better than the stage performance due to the lower pressure at the second sidestream flange. The second section will show inferior performance since the inlet flange total pressure being higher than the stage inlet total pressure and discharge flange total pressure is now lower than the stage discharge total pressure. The inferior performance of section 2 is exaggerated by a shift to reduced flow, relative to the stage curve, due to the higher inlet pressure which results in a reduced inlet volume flow.

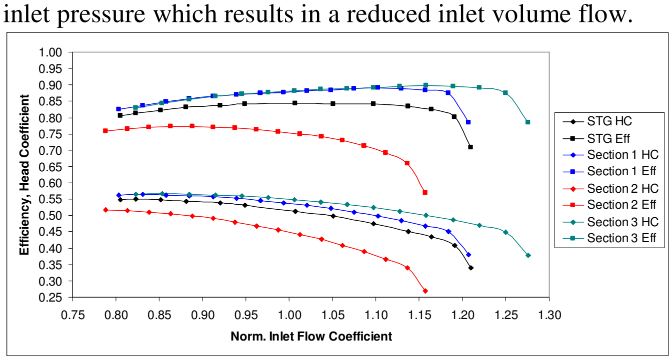

Figure 4. Sectional Performance: LC = 4 for SS1; LC = -4 for SS2

For scenario 3, the first sidestream has a negative LC (-4) and the second sidestream has a positive LC (4). The subsequent sectional performances are shown in Figure 5. In this case, the first and third sections will show a lower performance. The former due to lower discharge flange total pressure and the latter due to higher discharge flange total pressure. The performance for section 2 would be deemed by most observers to be unrealistically high as the efficiency is above the proven efficiency of centrifugal compressors. This unrealistic performance is due to inlet flange having a lower total pressure than the stage and the discharge flange having a higher total pressure. The impact is exaggerated by a shift to increased flow, relative to the stage curve, due to the lower inlet pressure which results in an increased inlet volume flow. The other effect is also increased as the flange losses are a greater percentage of the stage head due to lower head values in overload and the increase in dynamic pressure.

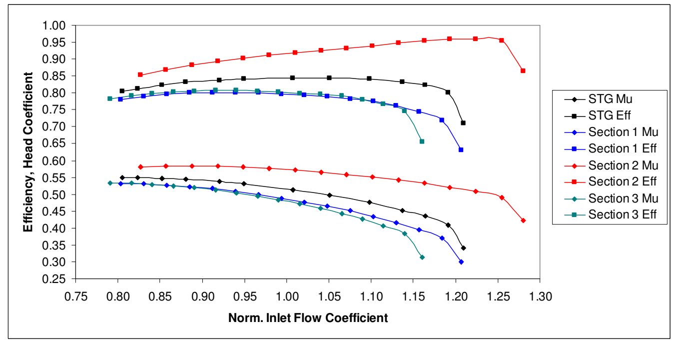

Figure 5. Sectional Performance: LC = -4 for SS1; LC = 4 for SS2

The final scenario highlights the sectional performance resulting from both sidestream possessing negative LCs (-4), Figure 6. In this case, the sectional performance will be opposite to that shown by Figure 6, where both sidestreams had positive LCs. Section 3 will show the highest performance, followed by section 2 and then by section 1. The lower flange total pressures at the sidestream are the reason for this result.

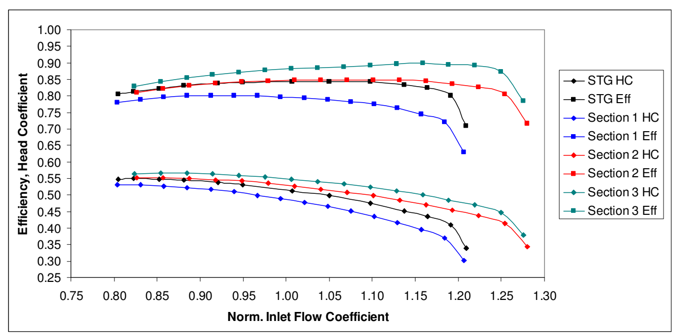

Figure 6. Sectional Performance: LC = -4 for both SS

Effect of Flow Ratio on Sectional Performance

Consider the same example machine consisting of three stages and two sidestreams constituting a three section machine. If the sidestream geometry and flow ratio is consistent for both sidestreams, then the sectional performance would be consistent. Next, the effect of varying the flow ratio at either sidestream on the performance of sections 1 and 2 will be shown.

For this example, the flow ratio at each sidestream was adjusted, after accounting for pressure losses due to geometry, such that each stage was operating at its design flow coefficient. At this flow ratio, and for the assumed geometry, both sidestreams possessed a negative loss coefficient. The subsequent sectional performances at these conditions, is shown by Figure 7.

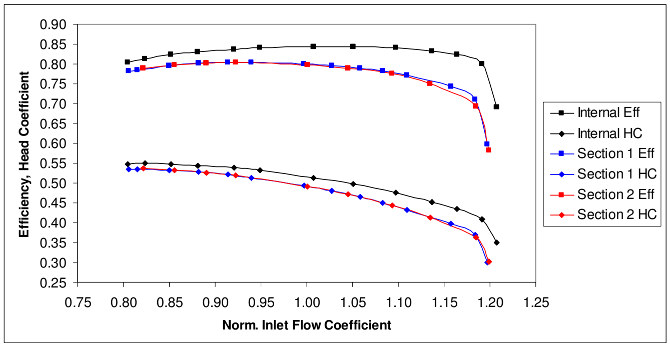

Figure 7. Sectional Performance at constant Flow Ratio

Consider the first section in this setup. If the flow ratio is varied between 90% and 110%, which are the tolerances as specified in the ASME code (PTC-10), the flange pressure changes accordingly and results in a change in the first section performance. This variance in performance is shown in Figure 8. At the design point (Flow coefficient = 0.277), the sectional efficiency varies between ± 2%. This variance reduces as the flow is decreased towards surge. However, as the flow is increased towards overload, the variance increases up to approximately ± 8%.

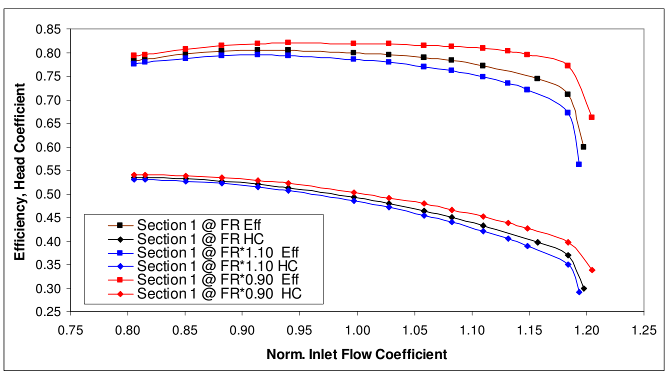

Figure 8. Section 1 performance variation when SS1 flow ratio varied at ±10%.

These differences in flange total pressure and flow contribute to fluctuations in the sectional efficiencies in the subsequent sections. Further, if the second sidestream is also not maintained at constant flow ratio, but allowed to fluctuate within a tolerance of ±10%, the effect on the sectional performance is compounded even further. Figure 9 shows the change in performance in section 2, when the first sidestream is at 90% flow ratio and the second sidestream is varied at ±10% of flow ratio. It can be seen that when the second sidestream operates at the lower limit of flow ratio (90% FF), the sectional performance is similar to what it would be if both sidestreams were operating at the design flow ratio. However, when the second sidestream operates at the upper limit (110% FF), at design flow the sectional efficiency drops by approximately 4%, and at overload it drops by approximately 9%.

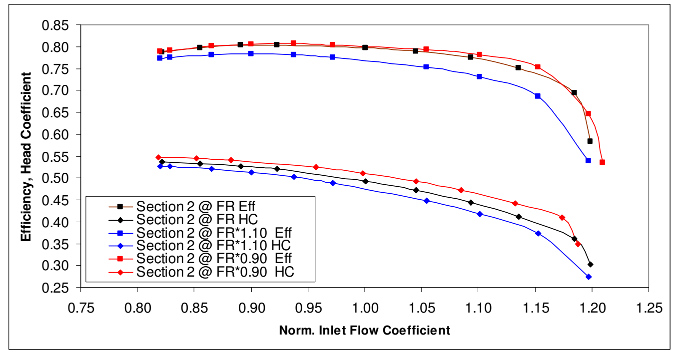

Figure 9. Section 2 Performance Variation @ ±10% FF when Sec 1 @ 90% FF

On the other hand, Figure 10 shows the variance in performance of Section 2 when the first sidestream is at 110% flow ratio, and the second sidestream is varied at ±10% of flow ratio. In this case, when the second sidestream operates at the upper limit of flow ratio (110% FF), the sectional performance is similar to what it would be if both sidestreams were operating at the design flow ratio. However, when the second sidestream operates at the lower limit (90% FF), at design flow the sectional efficiency increases by approximately 4%, and at overload it increases by approximately 9%.

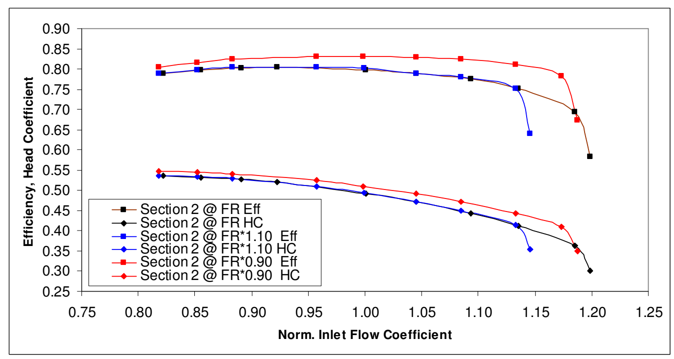

Figure 10. Section 2 Performance Variation @ ±10% FF when Sec 1 @ 110% FF

The greatest effect of varying the flow ratio will be on the performance of the second section, since it has a sidestream upstream and downstream of the stage. The effect on the performance of the third section depends on the loss coefficient of the second sidestream. A positive loss coefficient will result in a sectional efficiency and head lower than the stage performance. Conversely, a negative loss coefficient will show the sectional performance to be better than the stage performance.

Other than affecting the sectional performances, it is also important to note that variance in flow ratio impacts the velocity levels where the two streams merge. Significant variation in the velocity profile upstream of the impeller changes the incidence on the blade leading edge of the following impeller. This change in incidence leads to a change in the stage (internal) performance. This is a further cause of variation not contemplated in the previous examples.

INVESTIGATIVE STUDY

The previous sections have reviewed the sensitivity of sectional performance to flange losses and flow ratios, but the examples were simplified for illustrative purposes. In a real world environment many of the operating parameters that were fixed are variables which increase the difficulty of prediction. During testing and during field operation, it is often not possible to maintain the design flow ratio of the sidestream. In this situation analytical tools must be used to extrapolate the measured test results to predict these conditions.

In an effort to quantify the accuracy of the CFD and 1D prediction tools an investigation on an existing sidestream design was conducted by the OEM. The 1D model investigation was designed to quantify the accuracy when predicting flange to flange curves when provided an internal performance curve measured on test. The goal of the CFD study was to measure the ability of CFD to accurately predict both the internal and external (flange to flange) performance. Validation of both tools would be achieved by rig testing at design conditions and at off design conditions, including alternate flow ratios.

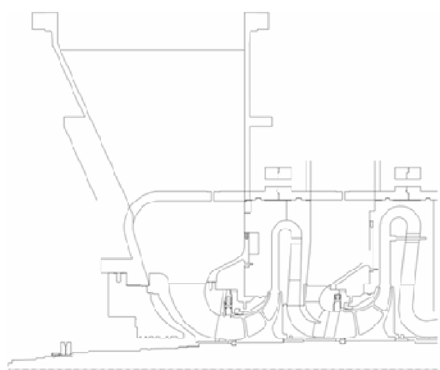

Figure 11. Sidestream Test Geometry Layout

The geometry (Figure 11) selected for the investigation consisted of two stages with a sidestream entering after stage one. Table 2 shows the specifications at which the test was run.

Stage	Flow Coefficient	Tip Mach Number	Flow Ratio	MW
1	0.10	1.15	0.708	102
2	0.11	1.15		102

Table 2. Test Specifications

PREDICTION METHODOLOGY

The 1D prediction methodology used to predict flange-to-flange performance for the first section was broken down into three parts, a) main inlet loss estimation, b) stage performance prediction (internal) and c) sidestream loss estimation. The main inlet losses are predominately friction losses and have been validated previously using CFD analysis (Koch et al. 1995, Michelassi and Giachi 1997) but can also be provided by test measurement. The internal performance prediction was provided by CFD during model development, but the intention of this exercise was to develop a model based on an internal test curve. The sidestream loss estimation was completed using the loss model previously published (Koch et al. 2011). Real gas models were used to calculate all performance data, i.e. total/static pressures and temperatures, gas velocities, gas densities and flange-to-flange efficiency and head.

Sectional Performance Calculations

The sidestream sectional performance prediction model requires the main inlet total conditions (pressure, temperature and mass flow) and flow ratio to determine stage and sectional performance. The model uses real gas calculations to generate total and static data (pressure, temperature, gas density, velocity, flow) at all key locations, i.e. main inlet flange, stage inlet, stage exit, sidestream exit, sidestream flange, second stage inlet, second stage discharge.

At the sidestream flange, the mass flow rate and the total temperature of the sidestream flow are controlled during testing. The flow ratio is kept consistent with the testing procedure in order to obtain comparable results. The sidestream exit total pressure (and conversely the flange total pressure) is a function of the sidestream flow velocity and the previous section static pressure. Therefore, the sidestream inlet flange volume flow is a ratio of the flange total pressure (and compressibility at that pressure and temperature) and mass flow (Koch et al. 2011).

The total mixed temperature entering the second impeller is determined per section 3.5.5 of the ASME PTC-10 guidelines (1997), on a mixed mass enthalpy basis. The ASME code assumes that the sidestream flange pressure is equal to the inlet pressure of the first stage of the next section. In reality this is not true and so the mixed total pressure is determined on a mass-averaged basis using sidestream and return channel exit total pressures, which is also suggested by Hardin (2002). The same methodology could then be applied to any subsequent sections that are followed by a sidestream and in this way, the performance map of the full compressor can be determined.

As mentioned earlier, only the first section results are discussed in this paper in order to gauge the effect of varying inlet conditions and sidestream flow ratio on the performance of that section.

TESTING METHODOLOGY

As shown in Figure 12, the test rig was extensively instrumented at key locations. These included the main inlet flange, first stage diffuser and return channel exit, sidestream flange, second stage inlet, diffuser exit, and return channel exit and discharge flange. Instrumentation consisted of combo-probes (half-shielded thermocouple and Kiel-head pressure probe), individual Kiel-head pressure probes, total pressure rakes, 5-hole probes, static pressure taps and dynamic pressure transducers.

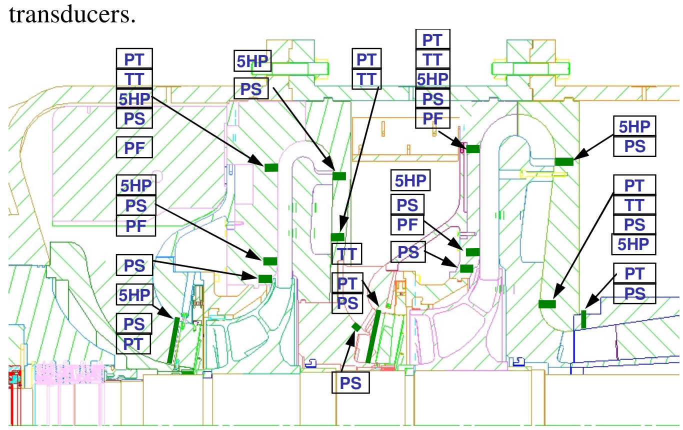

Figure 12. Instrumentation Layout of the Subscale Test Rig

The machine was tested using guidelines detailed in the ASME code. Per section 3.5.2, the ratio of the inlet volume flow to the section discharge flow (in this case the first stage return channel exit) must be kept within ±5% at the test conditions. For the sidestream flow, it is required that the ratio of the sidestream inlet flow to the discharge flow of the first section be kept within ±10%. During testing, this flow ratio was maintained within ±3% (see Kolata and Colby 1990). Section 3.5.3 of the ASME code specifies that the discharge temperature be measured prior to the mixing of the two flows, to prevent errors due to heat transfer effects between the two flows. Also, it is possible that any pressure measurements taken at the exit of the return channel may be affected by the mixing flow downstream. For this reason, the first section discharge flow ratio was calculated using the stage diffuser exit conditions. The sidestream flange temperature was maintained at the diffuser exit temperature. The test was conducted using R-134A refrigerant in order to achieve aerodynamic similitude at the full-scale design conditions.

As per section 3.5.5 of the ASME code, all performance calculations were done on a flange-to-flange basis using real gas evaluation. Each section was tested across the full range (i.e. from surge to overload), with about 10-13 stabilized points taken at appropriate flow intervals.

SIDESTREAM SECTION MODEL AND TEST DATA RESULTS

The test results for internal (stage) and external (sectional) normalized efficiency, normalized head coefficient and normalized sidestream flange pressure versus normalized flow coefficient are shown by Figures 13(a, b and c) respectively. All values are normalized using the design flow coefficient of the stage being used in the first section. For example, the tested sectional efficiency is normalized using the tested stage efficiency at design flow coefficient. It can be observed from Figs. 13(a) and 13(b) that although the internal stage curves for efficiency and head coefficient are smooth across the range, the corresponding sectional curves are not so. This is a direct result of the sidestream flange pressure curve not being smooth across the range as shown in Figure 13(c).

To validate the accuracy of the sidestream prediction model, the loss coefficient and stage performance curves from CFD were corrected per test results. This removes the CFD uncertainty as a source of variation. Using the main inlet conditions (mass flow, total pressure and total temperature), sidestream flange total temperature and flow ratio from test, the sidestream prediction model was then used to determine total/static pressure/temperature conditions at each location, i.e. main inlet, stage inlet, stage exit, sidestream exit and sidestream flange. The stage and sectional performance (efficiency, head coefficient etc) were determined using real gas thermodynamic calculations. The ensuing comparison is shown by Figures 14(a), 14(b) and 14(c). The sectional efficiency as predicted by the sidestream model agrees very well with the as-tested results, with the maximum deviation being 1.45% (Fig 14(a)). The sidestream flange pressure matches very well between that predicted by the sidestream model (Figure 15(c)) and test data. The maximum deviation here is within 0.6%. This validates the ability of the sidestream prediction model to accurately determine sectional performance for a given set of main inlet conditions and specified flow ratio from test.

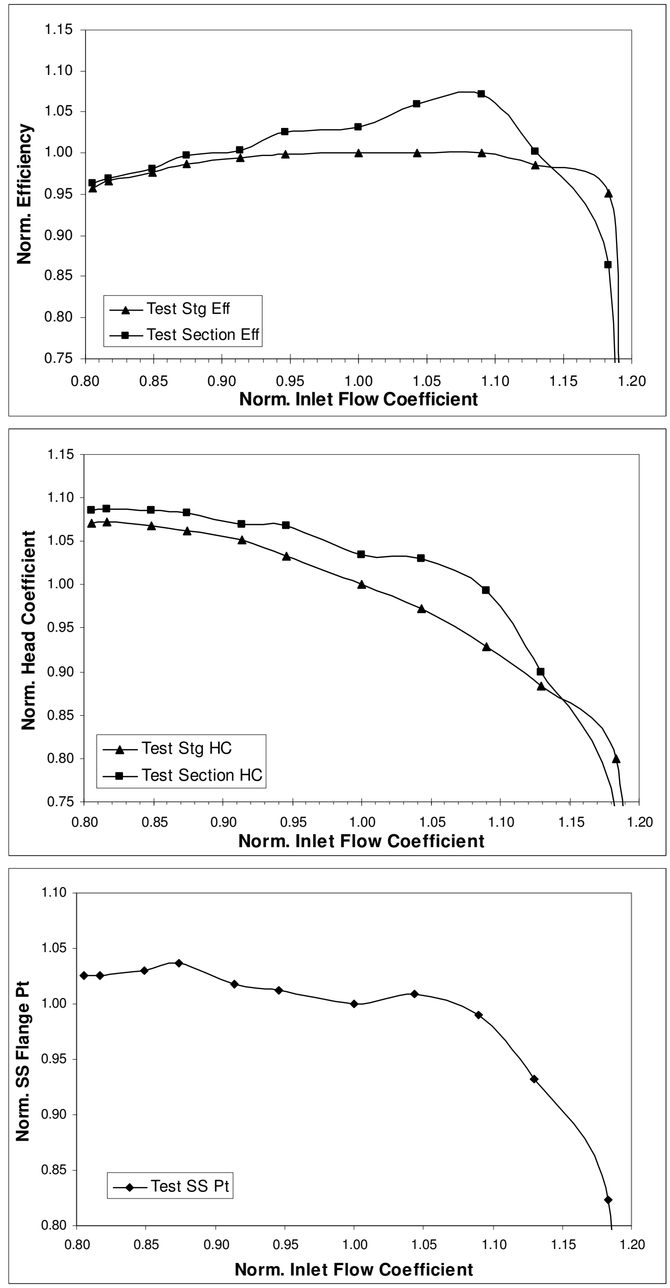

Figure 13. Test Results vs Norm. Phi for (a) Norm. Efficiency, (b) Norm. Head Coefficient, (c) Norm. Flg. Pt

Similar to the test results, the sidestream model (when using test validated conditions) also predicted non-smooth sectional efficiency and head coefficient curves. This was postulated to be due to two key reasons:

(i) Variation of the main inlet total pressure.
(ii) Variation of the flow ratio between sidestream flow and core flow

For the test, the total pressure at the inlet varied within ±2.5% of design inlet pressure and the flow ratio (sidestream to core flow) varied within ±3%. Note that both these tolerances are well within those specified by the ASME code under section 3.5.2.

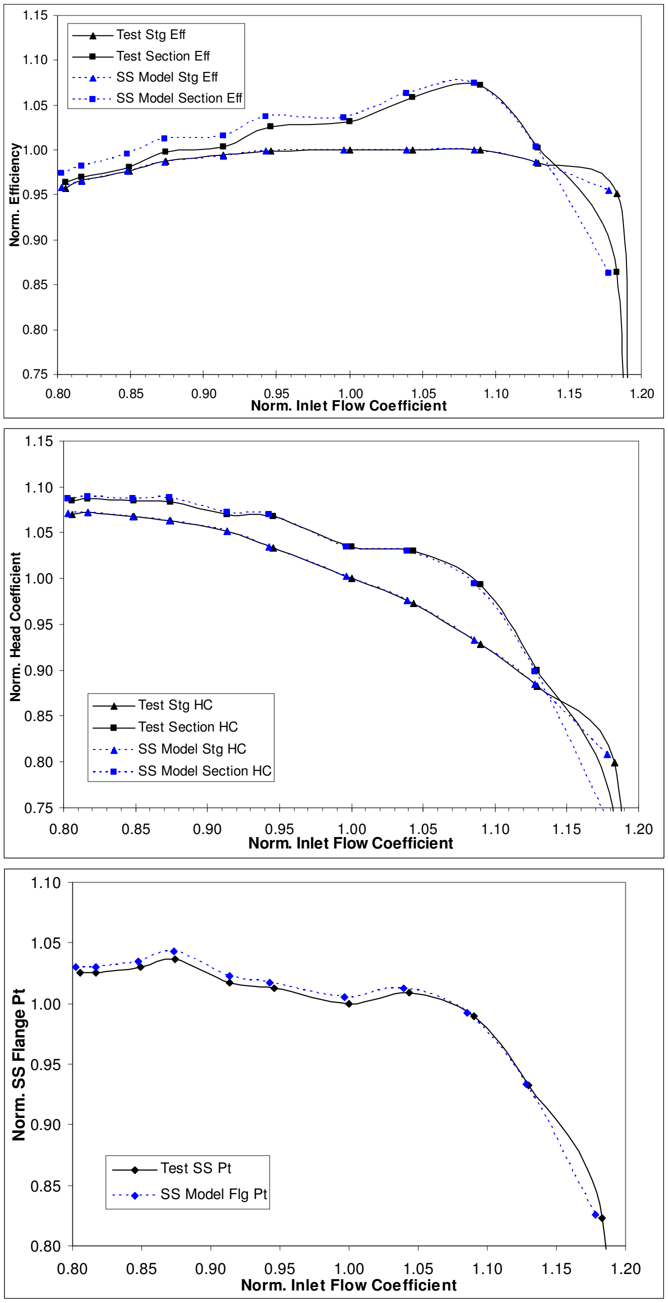

Figure 14. Model Prediction (Test) & Test Results vs Norm. Phi for (a) Norm. Efficiency, (b) Norm. Head Coefficient, (c) Norm. Flg. Pt

In order to eliminate the effect of these two factors, the sidestream model was re-run keeping uniform, consistent inlet conditions and maintaining the flow ratio within a tolerance of ±0.015%. The results of this run are shown in Figure 15. Keeping the above two factors uniform, it is shown that uniformly smooth sectional curves are indeed possible. However, we see from Figure 15 that the sectional efficiency is continuously rising from surge to overload. This effect is due to the constant flow ratio. In reality, as the compressor operates near overload the flow ratio will decrease since the increase in sidestream flange pressure cannot be maintained. In order for the flow ratio to remain consistent, the sidestream volume flow has to be increased accordingly. The static pressure of the sidestream flow at the mixing section is a direct ratio of the core flow static pressure and local curvature. However, the increased sidestream mass flow at the mixing section has to flow through a fixed area. This requires a larger flange pressure at the sidestream inlet, which is comparatively much larger than the total pressure at the return channel exit of the preceding stage. This is shown in Fig. 16, where as flow is increased, a gradually higher total pressure at the sidestream flange is required relative to the stage exit total pressure (delta Pt), in order to maintain the flow ratio. Since sectional efficiency is calculated based on flange pressures, an unrealistically high sectional efficiency is shown near overload for sidestreams when the flow ratio is constant – and in many cases, the sectional efficiency can exceed 100%. It is also noticed from Fig. 15(c) that as the flow is increased at constant flow ratio (sidestream to core flow), the sidestream flange total pressure is decreasing. Since, for many machines sidestream flows originate from processes at a fixed pressure, a fixed flow ratio across the flow range may not be desirable.

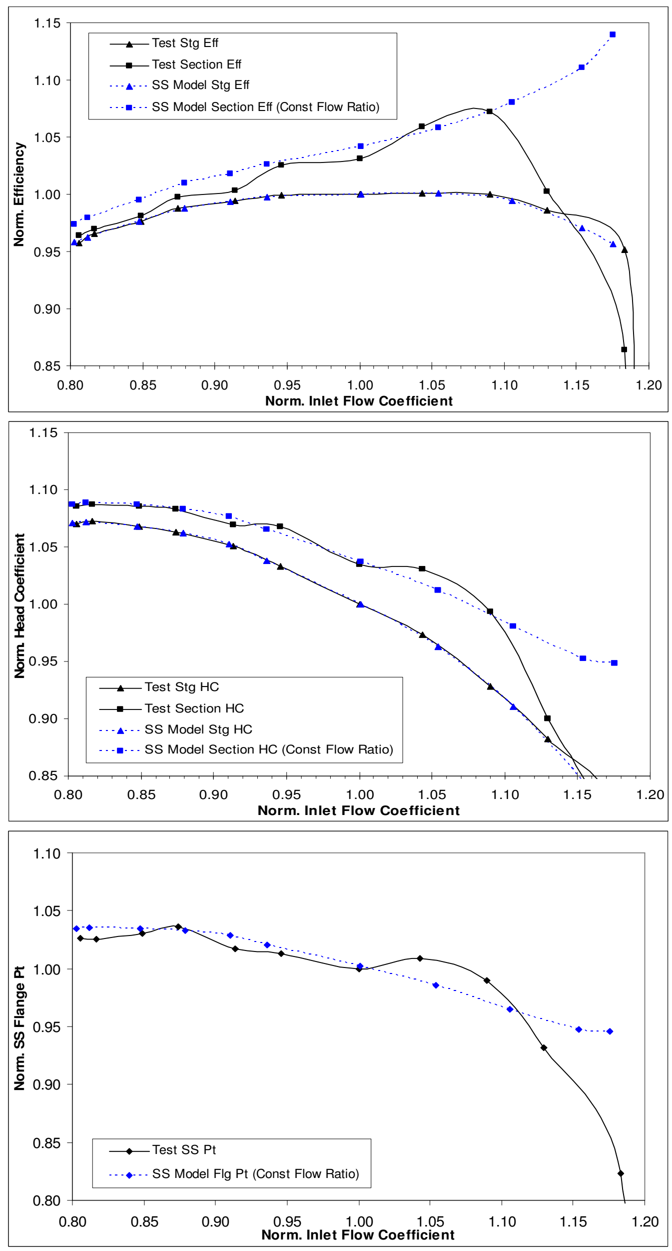

Figure 15. Model Prediction (Uniform Conditions) & Test Results vs Norm. Phi for (a) Norm. Efficiency, (b) Norm. Head Coefficient, (c) Norm. Flg. Pt

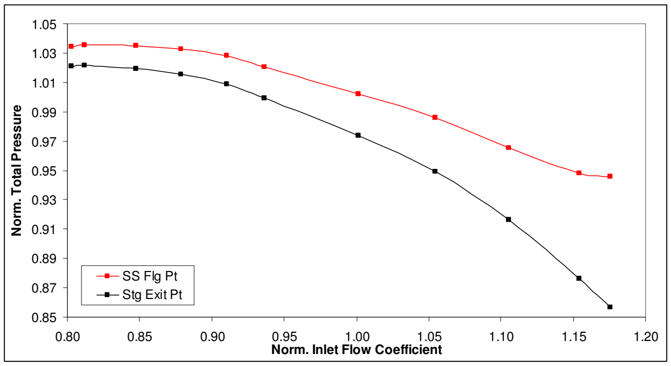

Figure 16. Total Pressure vs. Norm. Phi for Stage Exit Total Pressure and Sidestream Flange Total Pressure

In order to study the effect of varying the main inlet total pressure on the sidestream flange total pressure, the sidestream model was run by varying the main inlet total pressure within the range of ±2.5%. The effect of varying the inlet total pressure leads only to a flow shift of the operating condition and hence has no effect on the stage or sectional efficiencies and head coefficients. Figure 17 shows the effect of varying main inlet total pressure on the sidestream flange total pressure and is directly proportional to the main inlet total pressure, i.e., it fluctuates within the same range of ±2.5%. This implies that if testing/operators are not diligent with the inlet conditions, the resulting pressure at sidestream flange will not reflect what is required by the process.

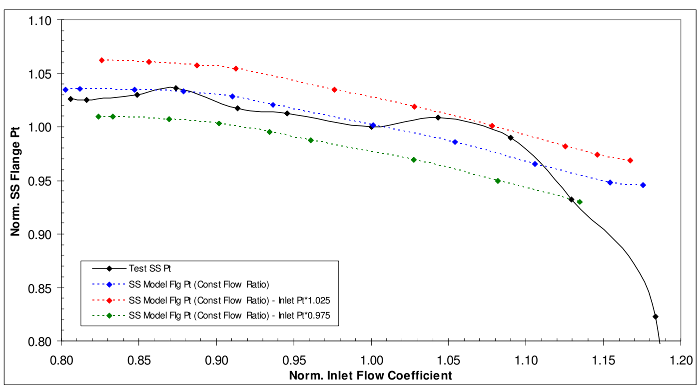

Figure 17. Model Prediction (Ptin ±2.5%) & Test Results vs Norm. Phi for Norm. Flg. Pt

ASME PTC-10 guidelines state that the allowable tolerance for the ratio of sidestream flow to core flow is ±10%. Figures 18(a), 18(b) and 18(c) show sectional efficiencies, head coefficients and sidestream flange pressures when the flow ratio is kept at design, increased by 10% and decreased by 10%. It can be seen by both Figures 18(a) and 18(b) that the stage efficiencies and head coefficients remain unchanged, i.e. the power consumption by the section is the same regardless of the flow ratio as expected. However, there are large discrepancies in the sectional efficiencies, head coefficients and sidestream flange pressures. In all three cases, the discrepancies increase in magnitude as the flow is increased from surge to overload. At the design point (normalized flow coefficient of 1), the sectional efficiency varies at ±6% (Fig. 18(a)). The head coefficient at the same design point varies at ±5% (Fig. 18(b)), whereas the flange pressure varies at ±3.7% (Fig. 18(c)). These errors get much larger as the flow is increased towards overload.

These results show that the variation in sectional performance due to flow ratio is highest at overload and lowest at surge. This implies that if the sectional performance is to be maintained within certain bounds during testing, the corresponding error in flow ratio needs to also be maintained within some bounds. If this is not done, the result will be a non-smooth curve as shown in Figure 13.

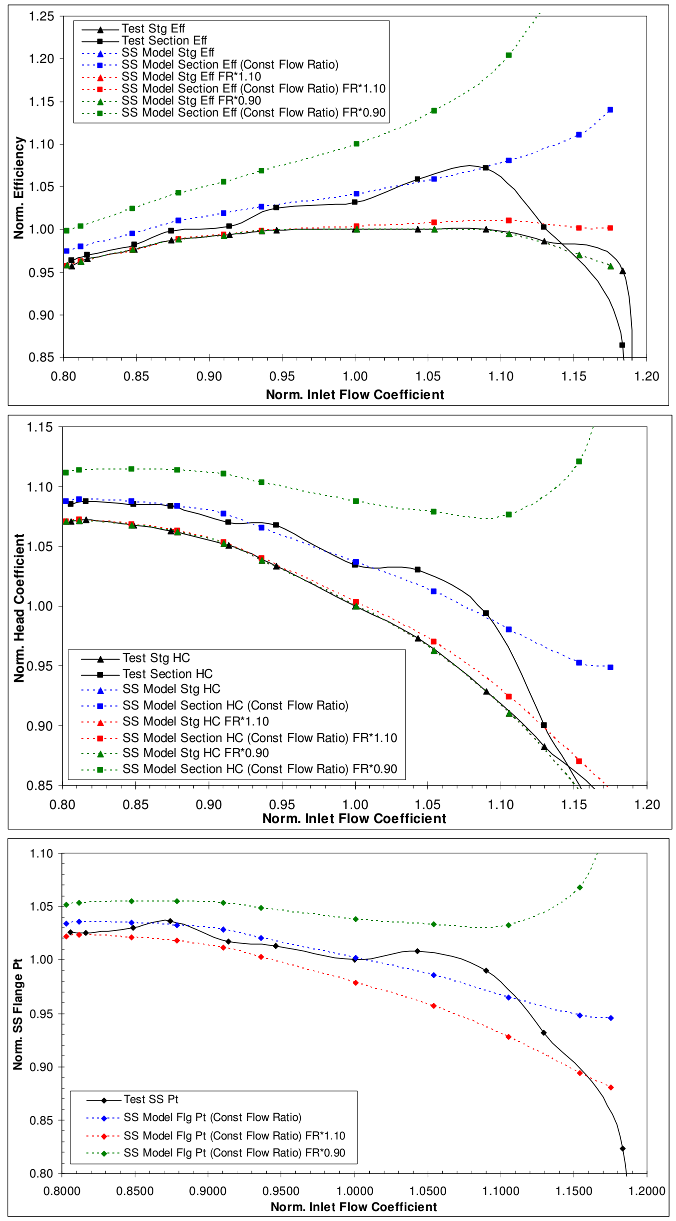

Figure 18. Model Prediction (Flow Ratio ±10%) & Test Results vs Norm. Phi for (a) Norm. Efficiency, (b) Norm. Head Coefficient, (c) Norm. Flg. Pt

The sidestream loss model was used to determine the allowable percentage error in flow ratio if sectional efficiency is to be maintained within ±1%. From Figure 19, it is shown that sectional performance is highly sensitive near overload. In order to maintain the sectional efficiency within ±1% variation across the flow range, the maximum allowable deviation in flow ratio is ±0.5%. For obvious reasons, this would be near impossible to achieve during testing and so some non-smoothness of the sectional performance curves for sidestream must be expected, especially near overload. At design flow, the maximum allowable deviation is 2.2% which is still well below the +/- 10% guideline. It is interesting to note that the allowable deviation near surge is similar to the ASME code tolerance of +/- 10%.

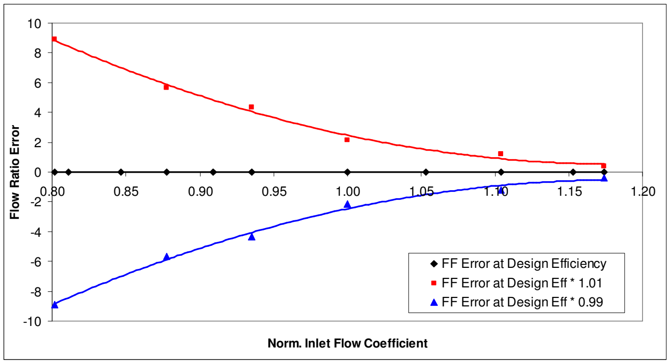

Figure 19. Allowable FF % Error for ±1% Sectional Efficiency Error

In summary, it is clear from the results of the sidestream prediction model that:

(i) Varying the inlet conditions has a direct impact on the sidestream flange pressure – of the same magnitude in terms of percentage as that of the change in inlet total pressure. However, the sectional efficiency and head coefficient are not affected.

(ii) Further, a variation in flow ratio has an even greater impact not only on the sidestream flange total pressure but also on the sectional efficiencies and head coefficients. The impact is greater as the flow is increased.

CONCLUSIONS

This paper proposed an overall model to determine sectional performance of centrifugal compressors with incoming sidestream flow. The proposed model is an extension of the work previously conducted by Koch et al. (2011), Sorokes et al. (2006) and Hardin (2002).

This paper demonstrated the variation in sidestream sectional performances due to loss coefficient and flow ratios using a fictional three stage, three section machine. It was shown that depending on the loss coefficient, the sectional performance can be better or worse than the stage performance. The effect of varying flow ratio on the various sections of the machine was also shown. The fluctuation in performance by varying flow ratios per ASME code (PTC-10) guidelines was compared with allowable API-617 and most client requested tolerances. It was found that ASME code (PTC-10) guidelines may not be sufficient to satisfy client expectations with respect to sidestream sectional performance.

The study also validated the sidestream prediction model using test data. The comparison of predicted data with tested data shows that sectional efficiency can be predicted within ± 1.5%, the sectional head within ± 0.5% and the sidestream flange total pressure within ± 0.6%. All of these tolerances are well within API-617 requirements and also within more stringent client requirements.

The impact of varying certain factors of flow ratio based on ASME code (PTC-10) guidelines was evaluated on the tested data. It was shown that, at design flow, varying the flow ratio (sidestream to core flow) within the ±10% as allowed in the ASME code (PTC-10) led to a variation in the sectional efficiency on the order of ±6%, variation in the head coefficient on the order of ± 5%, and variation of sidestream flange total pressure on the order of ± 4%. These variations increased as the flow increased (near overload), but decreased as the flow approached surge. At design flow, these variations are not only outside of the range required by some clients (sidestream pressure tolerances in the order of ± 2%), but also outside API-617 requirements (sectional efficiency tolerance of the order of ± 4%). These errors will be compounded for successive sections in a machine with several incoming sidestreams resulting in even larger deviations.

It is recommended that, in order to meet tolerances for sidestream flange pressures and sectional efficiencies, the requirement for flow ratio be made more stringent for sections with incoming sidestreams to at least ±5%, in order to stay within the bounds of acceptable variation in sectional efficiencies. As shown in Figure 19, even with these stricter requirements, it will still be difficult to generate smooth sectional performance curves during testing. It is suggested that post-test, the resulting performance curves be corrected using the sidestream prediction model and these corrected curves be used to determine if the client requirements have been achieved. Sidestream machines used for processes (such as LNG refrigeration processes) require fixed sidestream flange pressures. As shown by the sidestream prediction model, it is not possible to achieve this by keeping the flow ratio (sidestream to core flow) constant. The flow ratio will have to be adjusted according to the inlet flow at which the machine is operating in order to get the required corresponding sidestream flange pressure.

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NOMENCLATURE

Pt_{SS}	=	Total pressure at SS flange
Pt_{STG}	=	Total pressure at stage inlet
Pd_{SS}	=	Dynamic pressure at SS flange
Eff	=	Efficiency
HC	=	Head coefficient
FR	=	Flow Ratio
5HP	=	5 hole total pressure probes
PT	=	Total pressure probe
TT	=	Total temperature probe
PS	=	Static pressure taps
PF	=	Dynamic pressure probes

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